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RESEARCH MEMORANDUM

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ANALYTICAL INVESTIGATION OF FUEL-COOLED TURBINE BLADES WITH RETURN-FLOW TYPE OF FINNED COOLANT PASSAGES

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RESEARCH MEMORANDUM

ANALYTICAL INVESTIGATION OF FUEL-COOLED TURBINE BLADES WITH RETURN-FLOW TYPE OF FINNED COOLANT PASSAGES*

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SUMMARY

Coolant-flow rates for a turbine rotor blade with return-flow type of coolant-passage configuration formed by fins within a capped blade shell were investigated with both hydrogen and methane fuels as coolants. The investigation was made for a high-altitude supersonic turbojet engine (flight Mach number, 2.5; altitudes, 50,000 and 80,000 ft) with a turbine-inlet temperature of 3000° R. Ranges of coolant inlet temperatures from 100° to 1000° R and coolant inlet pressures from 3000 to 20,000 pounds per square foot were investigated.

When hydrogen fuel was used as the coolant, the coolant-flow requirements varied from 11 to 20 percent of the primary combustor fuel flow at a flight altitude of 50,000 feet and from 8 to 25 percent at a flight altitude of 80,000 feet. When methane fuel was used as the coolant, the coolant-flow requirements were 29 percent of the primary combustor fuel-flow requirements at a flight altitude of 50,000 feet. With regard to pressure drop, the problem of passing the required coolant-flow rates for this type of configuration did not appear serious. For equal pressure drop and duct length, the duct diameters when hydrogen was used as the coolant were as much as one and one-half times greater than those when methane was used. Cooling of gas turbines with fuels such as hydrogen and methane appears feasible.

INTRODUCTION

The use of liquid hydrogen or liquid methane as the fuel for a gas-turbine engine (refs. 1 and 2) would provide a large-capacity heat sink for cooling loads in high-speed aircraft. The purpose of this report was to investigate the use of these two fuels as coolants for a type of rotor blade in a high-temperature turbine.

At flight Mach numbers of about 2.5 and higher, it becomes increasingly difficult to utilize air bled from the engine compressor for

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turbine blade cooling because the cooling-air temperature is directly dependent on the flight speed and altitude. It is desirable, therefore, to find suitable methods of reducing the cooling-air temperature. This temperature can be reduced by (1) rejecting heat to a colder fluid by means of a heat exchanger, (2) using refrigeration systems such as those discussed in reference 3, or (3) spraying water into cooling air to make use of the latent heat of vaporization of the water. The specific liquid consumption required for the engine is increased, of course, if water is sprayed into the cooling air. Also, it may be difficult to provide a refrigeration cycle that is much better than using a heat exchanger alone (ref. 3). Fuels such as liquid hydrogen or liquid methane provide an excellent heat sink, so that a heat exchanger could be used with the fuel as the receiver. This system, as well as the other systems mentioned, however, results in added engine weight. Therefore, the use of the fuel as a coolant directly would be more desirable and would thus eliminate the weight and bulkiness of a heat exchanger.

There are two other important advantages to using fuel as a coolant. First, this type of turbine-cooling system is free from flight Mach number limitations because the fuel-coolant inlet temperature is largely independent of flight speed and altitude if the fuel tanks are insulated. Second, with regard to engine performance, rejecting the heat absorbed by the fuel coolant into the engine compressor-discharge air in the combustor is probably the most efficient turbine-cooling method known. The effects on engine performance are shown in reference 4 for a similar case where heat is rejected at the compressor exit for liquid-cooled turbines.

The use of fuel as a turbine blade coolant requires a blade that does not discharge the coolant into the gas stream but, instead, has a return-flow path in the blade so that the coolant (fuel) can be ducted to the engine burners after it cools the turbine blades. The feasibility of such a blade was investigated analytically with hydrogen as the coolant and is reported in reference 5. The blade configuration used (ref. 5) was about the simplest configuration that could be conceived for a return-flow blade. Other configurations could undoubtedly make more-effective use of the coolant.

Other factors that must be considered when fuel is used as a coolant are the stability of the fuel and any effects that the fuel may have on the turbine blade materials. Methane is one of the most stable of all the hydrocarbons; however, it does start to decompose at a temperature of 1530° R (850° K) (ref. 6). The bulk temperature of the methane would probably never reach this temperature, and the residence times would be very small for flow through a turbine blade. Fuel stability, therefore, will probably not be a problem, but experimental evaluation of the problem would undoubtedly be required. Using hydrogen as a coolant could possibly cause hydrogen embrittlement in turbine blades of some materials. With the proper material selection, this need not be a problem. Consequently, based on stability and effects on blade materials, both methane and hydrogen could possibly be used as coolants for turbine blades.

The purpose of this report was to investigate the possible use of engine fuels (hydrogen and methane) as coolants for turbine rotor blades and to determine the pressure-drop characteristics of these coolants for a turbine blade with a more effective coolant-passage configuration than that considered in reference 5 but for the same engine and flight conditions. The conditions of this investigation were for a high-altitude supersonic turbojet engine (flight Mach number, 2.5; flight altitudes, 50,000 and 80,000 ft) with a turbine-inlet temperature of 3000° R and a turbine blade-root centrifugal stress of 50,000 pounds per square inch. Coolant inlet temperatures from 100° to 1000° R were investigated.

BLADE DESCRIPTION

Since the coolant-passage configuration for the reverse-flow blade of reference 5 was not as effective as might be desired for distributing the coolant to the leading and trailing edges, a more-effective type of coolant passage, shown in figure 1(a), was used in the analysis reported herein. The hollow space inside the blade shell is divided into a series of spanwise passages by fins between the suction and pressure surfaces of the blade shell. Alternate fins have a gap between the end of the fin and the cap over the blade tip to allow the coolant flowing radially outward in one passage to cross over and return radially inward in the adjacent passage formed by the fins. Hereinafter that portion of the passage for radial-outward flow will be known as leg I of the coolant passage, while the portion for radial-inward flow will be known as leg II. The fins not only serve to form a series of passage pairs for outward and return flow of the coolant, but they also serve as augmenting heat-transfer surfaces within the blade shell.

For the major portion of the investigation, it was assumed that the fins were 0.005 inch thick and the fin pitch was 0.105 inch (0.100-in. spacing between fins). Since the distance across the blade varies from leading to trailing edges, a mean fin height of 0.200 inch was assumed. The analysis was made for a representative passage pair (leg I plus leg II), as indicated on figure 1(b). In order to determine the required coolant flow per blade, it was assumed that the blade could be represented by 12 such mean passage pairs placed side by side on its mean camber line. Coolant-passage-flow area was assumed to be constant with span. The effects of geometry on coolant-flow requirements were studied by changing the fin thickness to 0.010 inch and the fin height to 0.150 inch while keeping the fin pitch at 0.105 inch. The values of passage and fin geometry used are given in the following table:

Geometry	Average fin height, in.	Fin thickness, in.	Fin spacing, in.	Fin pitch, in.	Flow area per passage leg, sq in.	Hydraulic diameter, in.
A	0.200	0.005	0.100	0.105	0.0200	0.1333
B	.200	.010	.095	.105	.0190	.1288
C	.150	.005	.100	.105	.0150	.1200
D	.150	.010	.095	.105	.0143	.1164

ANALYSIS

The present analysis is concerned only with the turbine rotor blades. Equations were derived for the coolant-flow requirements of these blades with the following assumptions:

(1) The effective gas temperature is constant spanwise and chordwise and is equal to the midspan value.

(2) The gas-to-blade heat-transfer coefficient is constant chordwise.

(3) Coolant heat-transfer correlations for fully developed flow are used. (Passage inlet effects are ignored.)

(4) Radial and chordwise heat conduction in the blade shell is neglected. (Calculations using chordwise conduction indicated the effects of chordwise conduction are small.)

(5) Radiant heat transfer is neglected.

(6) The coolant-passage-flow area is constant in a spanwise direction.

(7) The 180° flow reversal in the coolant passage in the blade tip produces no pressure loss.

Basic Equations for Temperature Distributions

The equations used to calculate the average blade-shell temperature and the coolant temperature in the coolant passage for a segment of the span at a given spanwise station are derived in appendix B. (All symbols are defined in appendix A.) The average blade-shell temperature for the segment at a given spanwise station was calculated by means of the following equation:

$$T_{B,av} = \frac{h_o p T_{g,e} + h_{c,I} T_{c,av,I} \frac{p - \tau}{2} + h_{c,II} T_{c,av,II} \frac{p - \tau}{2} + k\tau\varphi \frac{\beta^2}{\varphi^2} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}}}{h_o p + h_{c,I} \frac{p - \tau}{2} + h_{c,II} \frac{p - \tau}{2} + k\tau\varphi \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}}} \tag{1}$$

where

$$T_{c,av,I} = \frac{T_{c,I,in} + T_{c,I,out}}{2} \tag{2}$$

and

$$T_{c,av,II} = \frac{T_{c,II,in} + T_{c,II,out}}{2} \tag{3}$$

Since the calculation of the blade-shell temperature distribution begins at the base of the blade, where the coolant inlet temperature is known, it is also necessary to know the coolant outlet temperature at the base. This coolant outlet temperature can be calculated from the total heat input into the blade:

$$\Sigma 4 h_o p_o (T_{g,e} - T_B)\Delta x = w_c c_p (T_{c,outlet} - T_{c,inlet}) \approx 4 h_o p b (T_{g,e} - T_{B,av}) \tag{4}$$

In order to calculate film temperatures on which to base coolant heat-transfer coefficients on the fin surface, the mean fin temperature was calculated from the following equation:

$$T_{f,av} = \frac{\beta^2}{\varphi^2} + \frac{T_{B,av} - \frac{\beta^2}{\varphi^2}}{\varphi L/2} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{5}$$

The coolant temperatures at a given spanwise position were calculated by the following two equations:

$$T_{c,I,out} = T_{c,I,in} + \frac{\omega^2 \Delta x}{2gJc_p} [2r_h + \Delta x(2n - 1)] + 2h_{c,I} \Delta x \frac{(p - \tau)(T_{B,av} - T_{c,av,I})}{w_c c_p} +$$

$$\frac{2h_{c,f,I} L \Delta x \left(\frac{\beta^2}{\varphi^2} - T_{c,av,I}\right)}{w_c c_p} + \frac{4h_{c,f,I} \Delta x \left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right)}{\varphi w_c c_p} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{6}$$

$$T_{c,II,in} = T_{c,II,out} + \frac{\omega^2 \Delta x}{2gJc_p} [2r_h + \Delta x(2n - 1)] - 2h_{c,II} \Delta x \frac{(p - \tau)(T_{B,av} - T_{c,av,II})}{w_c c_p} -$$

$$\frac{2h_{c,f,II} L \Delta x \left(\frac{\beta^2}{\varphi^2} - T_{c,av,II}\right)}{w_c c_p} - \frac{4h_{c,f,II} \Delta x \left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right)}{\varphi w_c c_p} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{7}$$

Basic Equations for Pressure Distribution

The basic equations used to evaluate the coolant pressure requirements are the same as those used in the investigation of reference 5 and are presented in reference 7. The solution of these equations was accomplished on an IBM 650 magnetic drum calculator.

ANALYTICAL PROCEDURE

The analytical procedures employed in this investigation were very similar to those employed in reference 5. The determination of the necessary coolant flow required the evaluation of the radial blade-shell temperature distribution by means of equation (1). This blade temperature distribution was obtained for an assumed coolant weight flow and then matched to an allowable spanwise blade temperature distribution. The matching process was a trial-and-error procedure of correcting the coolant flow until the calculated blade temperature was equal to the allowable temperature at some spanwise station but did not exceed it at any other spanwise station.

Before equation (1) could be solved to obtain a blade-shell temperature distribution which could be matched to an allowable temperature distribution, other factors or conditions needed to be known. These were: (1) turbine geometry and operating conditions, (2) allowable blade temperature, (3) gas-to-blade heat-transfer coefficients, (4) coolant heat-transfer coefficients, and (5) coolant friction coefficients. Item (5) was needed in the calculation of the pressure requirement. Of these, the first three conditions were identical to those used in the analysis of reference 5 and will not be discussed. The last two will be discussed in some detail, inasmuch as the geometry of the coolant passage and the heat-transfer and friction-factor correlations used were different.

Coolant Heat-Transfer Coefficient

Forced-convection heat-transfer correlations were used for the same reasons discussed in reference 5 under this heading. For the turbulent-flow regime, the heat-transfer correlation employed in the analysis of reference 5 was also used:

$$\mathrm{Nu} = 0.023\ \mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{8}$$

The turbulent-flow regime was assumed to exist for Reynolds numbers greater than 7000, as indicated by figure 2 of reference 8.

In the transition region between the laminar- and turbulent-flow regimes, the following heat-transfer correlation was used:

$$\frac{\mathrm{Nu}}{\mathrm{Re}} = 0.00313 \tag{9}$$

For the laminar-flow regime, the heat-transfer correlation used was dependent on the aspect ratio of the coolant passage, which is the ratio of fin height to fin spacing for the investigation reported herein. Except for the study of the conduction effects of fin height and fin thickness, the major portion of the analyses was for a coolant passage with an aspect ratio of 2. Therefore, throughout this investigation, the correlation for laminar flow was taken to be

$$\mathrm{Nu} = 3.39 \tag{10}$$

as read from the curve for constant wall temperature for an aspect ratio of 2 from figure 9 of reference 9.

The limiting Reynolds number between the laminar-flow regime and the transition-flow regime was determined from equations (9) and (10) to be about 1100. Fluid properties in these and the following equations were based on film temperatures. Fluid properties of gaseous hydrogen were obtained from reference 10, and fluid properties of gaseous methane were obtained from reference 11.

Coolant Friction Coefficients

In order to determine the pressure requirements for the type of blade coolant-passage configuration being analyzed, it was necessary that the coolant friction factors be known. For the turbulent-flow regime ($\mathrm{Re} > 7000$), the friction factor was calculated in the same manner as in reference 5 by means of the Kármán-Nikuradse equation:

$$\frac{1}{\sqrt{4f}} = 2\log_{10}\left(\sqrt{4f}\ \mathrm{Re}\right) - 0.8 \tag{11}$$

In the transition region, the friction factor was assumed to be constant:

$$f = 0.0085 \tag{12}$$

For fully developed laminar flow, the friction factor was

$$f = 15.6 \text{ Re} \tag{13}$$

which was determined from figure 17 of reference 9 for an aspect ratio of 2. The limiting Reynolds number between the laminar- and transition-flow regimes as determined from equations (12) and (13) was 1835.

Calculation Procedure

In order to determine the required coolant flow within the pressure-drop limitations for a given coolant, blade operating conditions, and blade geometry, the required coolant flow was determined by heat-transfer and strength-of-material requirements. This coolant flow was then used to determine the pressure requirements. The blade was divided into 10 equal spanwise segments, and the heat-transfer calculations were initiated at the blade root with an assumed coolant flow. (Trial calculations with the blade divided into 20 segments resulted in better accuracy of the results. However, the accuracy with 10 segments was considered adequate.) The average blade temperature T_B (eq. (1)) of each succeeding segment was calculated by an iterative procedure. Since equation (1) represents conditions in both legs of the coolant passage, it was necessary to consider both legs in the calculation for a segment.

The segment inlet coolant temperature in leg I, the segment outlet coolant temperature in leg II, and the allowable blade temperature were used in an initial evaluation of the segment average blade temperature (eq. (1)). Substituting this segment blade temperature in equations (6) and (7) resulted in segment coolant outlet and inlet temperatures for legs I and II, respectively. Average coolant temperatures were then determined from equations (2) and (3), and the segment average blade temperature was recalculated using these average coolant temperatures and the previously calculated T_B to determine film temperatures for the heat-transfer coefficients. Initiation of the procedure at the blade root required the evaluation of the coolant outlet temperature for leg II. This temperature was approximated from equation (4) assuming the total heat input to the blade was determined by substitution of the allowable temperature at the midspan of the blade for the unknown average blade temperature.

The calculated temperature distribution for the complete blade was compared with the allowable blade temperature distribution. Appropriate corrections were applied to the coolant flow in successive trials until the calculated temperature at some point on the span very nearly agreed (within $\pm 1^{\circ}$ R) with the allowable temperature but did not exceed it at any other spanwise station. The heat flow as expressed by the summation of equation (4) was then compared with the assumption of heat flow mentioned previously. The entire procedure was then repeated, if necessary, for a corrected coolant flow to satisfy equation (4). The final value of coolant flow was used in the pressure distribution equations to determine the pressure changes in the passage.

All calculations were made on an IBM 650 magnetic drum calculator.

Range of Conditions

The engine operating conditions are as follows:

Engine compressor weight flow at flight altitude of 50,000 ft, lb/sec	289.7
Engine compressor weight flow at flight altitude of 80,000 ft, lb/sec	68.1
Turbine-inlet temperature, $^{\circ}$R	3000
Turbine blade tip speed, ft/sec	1425
Turbine tip diameter, ft	2.74
Hub-tip radius ratio	0.635
Blade length, ft	0.50
Number of turbine blades	50

For each coolant (hydrogen and methane) the required coolant flow was calculated for the allowable blade temperatures, which were based on stress-to-rupture data for the high-strength alloy A-286 at stresses equal to one and one-half times the centrifugal stress (centrifugal stress at the blade root, 50,000 psi). Thus, the stress-ratio factor was 1.5. The conditions over which each coolant was investigated are as follows:

Hydrogen:

Flight altitude, ft	50,000, 80,000
Effective gas temperature, $T_{g,e}$, $^{\circ}$R	2765
Gas-to-blade heat-transfer coefficient at flight altitude of 50,000 ft, h_o, Btu/(sec)(sq ft)($^{\circ}$R)	0.058
Gas-to-blade heat-transfer coefficient at flight altitude of 80,000 ft, h_o, Btu/(sec)(sq ft)($^{\circ}$R)	0.0213
Coolant inlet temperature, $T_{c,inlet}$, $^{\circ}$R	100, 250, 500, 750, 1000
Coolant inlet pressure at flight altitude of 50,000 ft, $P_{c,inlet}$, lb/sq ft	3000, 6000, 9000, 20,000
Coolant inlet pressure at flight altitude of 80,000 ft, $P_{c,inlet}$, lb/sq ft	3000, 6000, 9000

Methane:

Flight altitude, ft 50,000
Effective gas temperature, $T_{g,e}$, ^{o}R 2765
Blade outside-surface heat-transfer coefficient, h_o,
  Btu/(sec)(sq ft)(^{o}R) 0.058
Coolant inlet temperature, $T_{c,inlet}$, ^{o}R 250, 500
Coolant inlet pressure, $P_{c,inlet}$, lb/sq ft . . 3000, 6000, 9000, 20,000

RESULTS AND DISCUSSION

When fuel is used as the turbine coolant, the factors that are important to know are (1) how the required coolant flow compares with the required fuel flow, (2) what pressure losses can be expected in the coolant system, (3) what is the fuel temperature rise, and (4) what size ducts should be used for the various fuels.

The heat transferred to the fuel is a function of the required blade temperature, the effective gas temperature, and the gas-to-blade heat-transfer coefficient; but the coolant temperature rise is also a function of the flow rate and the specific heat of the fuel. As long as the required coolant flow is less than the required fuel flow, the temperature rise of that portion of fuel being used as a coolant can be controlled to some extent by varying the cooling effectiveness of the blade. A blade with a high cooling effectiveness will require a smaller amount of coolant than a blade with lower cooling effectiveness, and the coolant temperature rise will be higher. It is possible, therefore, that a high cooling effectiveness in the blade may not be desirable in order to keep the fuel temperature rise at reasonable values, particularly if the fuel is unstable at high temperatures. With the higher coolant-flow rates, however, it is possible that trouble may be encountered due to excessive pressure losses. For a particular application, therefore, compromises between coolant-flow rate, pressure losses, and temperature rise may be required.

Inasmuch as it is improbable to make generalizations with respect to specifying the best compromise for all applications, no attempt will be made to do so herein. Instead, trends will be shown to illustrate the relative merits of hydrogen and methane as coolants for the blade configuration shown in figure 1. No attempt will be made to determine an optimum coolant-passage configuration. The cooling effectiveness of this blade will also be compared with the reverse-flow blade analyzed in reference 5.

Spanwise Blade and Coolant Temperature Distributions

Typical spanwise temperature distributions for the blade shell and for the coolant are shown in figure 2 for the highest and lowest coolant temperatures investigated for both hydrogen and methane. For the range of conditions investigated, the blade-shell temperature distribution was quite flat. The greatest variation from root to tip is shown in figure 2(c) to be about 90° R, which is for a methane coolant inlet temperature of 250° R at a flight altitude of 50,000 feet. For the methane coolant inlet temperature of 500° R, the shell temperature variation is the least shown (about 15° R). Other conditions investigated for hydrogen but not shown in figure 2 had practically no spanwise variation. All the blade-shell temperatures matched the allowable at the root.

For all the conditions investigated except that shown in figure 2(a) (solid line, coolant inlet temperature of 100° R), the temperature rise in leg I was greater than in leg II. The temperature rise was generally less in leg II because the coolant had become warmer and the temperature differences between surfaces and coolant were less. Consequently, less heat was transferred to the coolant in leg II provided the heat-transfer coefficient was not radically different between legs I and II. The exception mentioned is an example of a case where the heat-transfer coefficients were very much lower in the first few segments of leg I, because the flow was in the laminar regime and shifted to the transition-flow regime toward the tip of leg I. Flow in leg II was all in the transition-flow regime, and the coefficients were considerably higher. The coolant temperature rise in both legs I and II for this type of return-flow blade was quite unlike the coolant temperature distribution in legs I and II of the return-flow blade of reference 5. In that blade, the coolant in the return-flow leg had no contact with the outside shell (coolant flowed radially inward through an insert inside the coolant cavity, see fig. 1(c)); therefore, the highest coolant temperature was at the blade tip and the coolant temperature decreased in the return-flow leg.

In all cases investigated, the highest coolant inlet temperature resulted in the highest coolant outlet temperature. Figure 2(b) shows the highest coolant outlet temperature for hydrogen to be 1375° R at a flight altitude of 80,000 feet for a coolant inlet temperature of 1000° R. Figure 2(c) shows the highest coolant outlet temperature for methane to be 1130° R for a coolant inlet temperature of 500° R. This temperature is still well below the decomposition temperature of 1530° R for methane.

As previously observed, for the range of conditions investigated in this report, the calculated blade-shell temperature distribution was relatively flat and was matched to the allowable temperature at the blade root. A value of the heat transferred to the blade, closer to the final value, could probably have been obtained if the allowable blade temperature at the blade root (instead of at the midspan) had been used in equation (4) to calculate the coolant outlet temperature for an assumed weight flow. Thus a trial-and-error solution for the required coolant weight flow applied to the first segment only could have resulted in a value closely approximating the required coolant flows reported herein.

Effects of Coolant Inlet Temperature on

Hydrogen-Coolant-Flow Requirements

The variation of required hydrogen coolant flow per blade with coolant inlet temperature is presented in figure 3 for two flight altitudes and a flight Mach number of 2.5. For the altitude of 50,000 feet, the required coolant flow reaches a minimum of 0.00725 pound per second at a coolant inlet temperature of about 350° R. The trends with temperature are similar to those reported in reference 5. The discussion in reference 5 with respect to the factors leading to these trends (e.g., the minimum for transition flow at an altitude of 50,000 ft) applies to the trends indicated in figure 3. For the altitude of 80,000 feet, such a minimum does not occur, because the flow regime shifts from transition at the higher temperatures to laminar flow for temperatures below 650° R.

Comparison of Return-Flow-Blade Coolant Requirements

Figure 4 shows a comparison of the coolant-flow requirements for the blade considered in this report (fin spacing, 0.100 in.; fin thickness, 0.005 in.) and the return-flow blades with insert of reference 5. Coolant flows are shown for three different spacings between the blade shell and the insert. The coolant-flow requirement for the blade with fins is approximately the same as that for the insert blade with the smallest spacing between the shell and the insert and is better than that of the insert blade with larger spacings. However, difficulty could be encountered in passing the required quantity of coolant through the blade with a spacing of only 0.020 inch (ref. 5). The results shown in table II, to be discussed in the section "Pressure requirements," show that this difficulty was not encountered in the blade with the fins. Therefore, when both heat transfer and pressure losses are considered, the blade with fins is more effective than the blade with the insert. As mentioned previously, however, it may not always be desirable to utilize the most effective blade. In some cases it may be better to use a blade which requires a larger coolant flow in order to keep the coolant temperature rise at a reasonable value.

Effects of Fin Thickness and Fin Height

The results of the study of the effects of changing the fin thickness and the fin height are presented in table I. This study was made for the hydrogen coolant at a flight altitude of 50,000 feet and a coolant inlet temperature of 250° R. From these results it can be seen that, for a fin spacing of 0.100 inch, decreasing L/τ (ratio of fin height to fin thickness) from 40 to 30 reduced the required coolant flow by about 14 percent. Similarly, for the 0.095-inch spacing, decreasing L/τ from 20 to 15 decreased the required coolant flow by about 11 percent. Evidently, a more optimum geometry than that used (fin spacing, 0.100 in.; L/τ, 40) in the major portion of the investigation could have been selected.

Comparison of Hydrogen and Methane as Coolants

As mentioned in the INTRODUCTION, liquid hydrogen and liquid methane seem to be feasible fuels to use as coolants because of the high heat absorption potential that they possess by virtue of their very low temperature when stored in the liquid state and their high specific heat in the gaseous state. The critical temperature and pressure for hydrogen are 59.9° R and 2714 pounds per square foot, respectively; for methane, the critical temperature and pressure are 343.2° R and 97,000 pounds per square foot, respectively. At 1 atmosphere of pressure, hydrogen boils at 36.7° R and methane boils at 201.1° R.

As shown in reference 5, the coolant-flow rate required is almost inversely proportional to the specific heat of the coolant. In the temperature range encountered (100° to 1000° R) in cooling the turbine blades, the average specific heat for hydrogen is about 3.5 Btu per pound per °R. Over the temperature range (250° to 500° R) investigated for methane, the average specific heat of methane is only 0.5 Btu per pound per °R. It would be expected, therefore, that the quantity of hydrogen required for cooling would be considerably less than that for methane; but, since hydrogen has a heating value over twice that of methane (51,623 and 21,520 Btu/lb, respectively), the hydrogen fuel-flow rate is also less.

A further consideration is the duct sizes required for hydrogen and methane for transporting the coolant to and from the turbine. Even though the weight-flow quantity is smaller for hydrogen, its density is so low that the duct sizes are not necessarily smaller, as shown in reference 5 in a comparison of duct sizes for air and hydrogen. These effects for hydrogen and methane will be illustrated in the figures showing the results of the analysis reported herein.

Fuel-coolant-flow requirements. - Figure 5 presents the turbine rotor-blade fuel-coolant-flow requirements for both methane and hydrogen in terms of fuel-to-engine-air weight-flow ratios so that they can be easily compared with the primary combustor fuel-flow requirements (also presented as fuel-air ratios) for a turbine-inlet temperature of 3000° R. The flight Mach number was 2.5. The comparison is presented for a wide range of coolant inlet temperatures (100° to 1000° R) for hydrogen. For methane, the coolant inlet temperatures should be kept low enough so that the coolant is not heated to the point where it may become unstable. Consequently, methane coolant-flow rates were only calculated for two points, both at low temperatures (250° and 500° R).

With methane fuel as the coolant (given only for the altitude of 50,000 ft), the coolant-flow requirement is about 29 percent of the primary combustor fuel-flow requirement. When hydrogen fuel is used as the coolant, the required coolant-flow rate for the rotor blades varies from 11 to 20 percent of the primary combustor fuel-flow rates at the 50,000-foot altitude. At the 80,000-foot altitude, the hydrogen fuel-coolant-flow rates vary from about 8 to 25 percent of the primary combustor fuel flow. Cooling the turbine stator blades would probably require approximately the same amount of coolant as for the rotor blades. It is indicated, therefore, that the fuel-flow rate to the combustor is ample for cooling both the rotor and stator blades for either hydrogen or methane. As expected, because of the high specific heat of hydrogen, hydrogen is a much better heat sink (larger heat-absorption capacity) than methane.

Pressure requirements. - The results of the pressure-requirement calculations for hydrogen and methane are given in table II. The case at a Mach number of 1.0 in the coolant passage is indicated by "no solution" in part (a) for hydrogen. This condition occurred for a coolant inlet temperature of 1000° R and a coolant inlet pressure of 3000 pounds per square foot. In a majority of the cases, there was a rise in the coolant static pressure to the blade tip in leg I, while in leg II the pressure dropped to a level that was still higher than the inlet static pressure. Thus a net pumping force was generated that could be utilized to overcome pressure drop in the ducting to and from the blade.

In the remaining cases, there was a rise in static pressure to the tip in leg I, but the pressure in leg II fell to a level lower than that to the inlet in leg I. This condition prevails for the cases with higher coolant inlet temperatures and lower inlet pressures at both altitudes. However, since more heat is transferred to the coolant at a flight altitude of 50,000 feet (because of higher gas-to-blade heat-transfer coefficient), this condition prevails for higher inlet pressures than it did at a flight altitude of 80,000 feet. For methane, this condition exists for the 500° R inlet temperature with an inlet pressure of 3000 pounds per square foot. A net pumping force is developed for the other methane cases. Thus, there does not appear to be a major problem in passing the required coolant flow through the type of return-flow coolant passage investigated in this analysis.

Coolant ducting size. - In reference 5 a method is derived for comparing the relative duct or pipe diameters required for the cases where the following values were constant: (1) the ratio of the pressure loss in the duct to the duct inlet pressure, (2) the pipe length, and (3) the coolant Mach number. The final equation for diameter is

$$D = K_3 \gamma^{1.25} (\mu/w_c)^{0.25}$$

where K_3 is a constant.

With the use of this equation, the relative pipe diameters were calculated for the required coolant-flow rates for hydrogen and methane at a flight altitude of 50,000 feet and a flight Mach number of 2.5. With the pipe diameter for methane at a coolant inlet temperature of 250° R as a base, the relative pipe diameters for hydrogen and methane are shown in figure 6 for a range of coolant inlet temperatures. The results for hydrogen and methane are similar to those shown in reference 5 for hydrogen and air. Even though considerably less weight-flow rate is required for hydrogen, the pipe diameter required will have to be over one and one-half times larger than that for methane. Larger diameters can often cause difficulty due to sealing problems, ducting weight, and turbine bearing size if the coolant is ducted through the turbine shaft. Specific engine applications would have to be considered, however, before it could be determined how detrimental the larger pipe sizes required for hydrogen might be.

Feasibility of Fuel-Cooled Turbines

With regard to both thermodynamics and heat transfer, the use of fuel for cooling the turbine seems entirely feasible. The coolant-flow requirements are less than the fuel-flow requirements; turbine cooling is not limited by flight Mach number; and, from a consideration of engine performance, rejecting heat removed by the fuel coolant into the engine compressor-discharge air in the combustor is probably the most efficient method of turbine cooling known. The only problems that occur, then, are practical problems in the fabrication of this type of coolant system, safety considerations, and the choice of suitable fuels that have a large heat capacity and are stable over the temperature range within which they must operate as coolants.

No insurmountable problems are involved in blade fabrication. The return-flow insert-type blades analyzed in reference 5 have been fabricated at the NACA Lewis laboratory. If the transverse fins of the blades analyzed herein are formed from corrugated sheet metal, the corrugated sheet can form the divider between the inlet and outlet flows with a crossover at the blade tip. Other types of reverse-flow blades are probably equally as feasible.

Perhaps the biggest problems will be the ducting and the seals between stationary and rotating parts. Ducting the fuel in and out of the turbine would undoubtedly be more difficult in a multistage turbine than in a single-stage turbine. With a single-stage turbine, the inlet could be on one face of the turbine and the discharge on the opposite face. Either shrouded or split-type wheels seem feasible. A simple ducting method is not apparent for a multistage turbine. It may be necessary to run two concentric ducts through the turbine shaft. If the duct sizes become large, bearing problems could occur with this arrangement. For very high flight speeds, however, the single-stage turbines would probably be adequate, and thus the problems with multistage turbines would be avoided.

The development of seals that would be completely free from leaks between the rotating and stationary parts of the coolant system appears to be remote. With the proper design, however, these leaks need not be a hazard. The areas around these seals could be ventilated to the primary combustor or the afterburner so that the fuel that leaked by the seals would be burned during engine operation. Upon shutdown, the engine would have to be vented so that pockets of gaseous fuel could not accumulate, and it would probably be desirable to purge the system with a noncombustible gas such as nitrogen or helium.

Although there would undoubtedly be problems in building engines with fuel-cooled turbines, none of the problems seems insurmountable. This type of cooling system therefore appears to be feasible.

CONCLUSIONS

Coolant-flow rates for a turbine rotor blade with a return-flow type of coolant-passage configuration formed by fins within a capped blade shell were investigated using hydrogen and methane fuels as coolants. The investigation was made for a high-altitude supersonic turbojet engine (flight Mach number, 2.5; altitudes, 50,000 and 80,000 ft) with a turbine-inlet temperature of 3000° R and a turbine blade-root stress of 50,000 pounds per square inch. Ranges of coolant inlet temperatures from 100° to 1000° R and coolant inlet pressures from 3000 to 20,000 pounds per square foot were investigated. The following conclusions were observed.

1. The blade with the coolant-passage configuration formed by fins within a capped blade shell can be cooled more effectively than a capped blade shell with insert for the same engine operating conditions.

2. The dimensions used for the investigation of this type of geometry were not optimized. The effectiveness of this type of coolant-passage configuration could be improved by using lower fin-height-to-thickness ratios than were used for the major portion of the investigation.

3. When hydrogen was used as the coolant as well as the fuel, the coolant-flow requirements varied from 11 to 20 percent of the primary combustor fuel-flow rate at a flight altitude of 50,000 feet and from 8 to 25 percent at a flight altitude of 80,000 feet.

4. When methane was used as the coolant as well as the fuel, the coolant-flow requirements were 29 percent of the primary combustor fuel-flow rate at a flight altitude of 50,000 feet.

5. For the wide range of coolant inlet temperatures and pressures investigated, the required coolant-flow rates for both hydrogen and methane were passed through the coolant-passage configuration with either a net pressure rise or a slight pressure drop. Thus, with regard to pressure drop, there did not appear to be a serious problem in passing the required coolant-flow rates for this type of configuration.

6. Of the two fuel coolants investigated, hydrogen required ducts whose diameters were one and one-half times larger than those for methane.

7. Cooling of gas turbine blades with fuels such as hydrogen and methane appears feasible.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, April 9, 1957

APPENDIX A

SYMBOLS

A	fluid free-flow area, sq ft
b	blade span, ft
C_1, C_2	constants in eq. (B3)
c_p	specific heat, Btu/(lb)(^{O}R)
D	diameter, ft
D_h	hydraulic diameter, ft
f	friction factor
g	acceleration due to gravity, 32.174 ft/sec^2
h	heat-transfer coefficient, Btu/(sec)(sq ft)(^{O}R)
J	mechanical equivalent of heat, 778 ft-lb/Btu
K_3	constant
k	thermal conductivity, Btu/(ft)(^{O}F)(sec)
L	fin height, ft
Nu	Nusselt number, hD_h/k
n	number of increments (1 to 10)
P	static pressure, lb/sq ft
Pr	Prandtl number, $c_p\mu/k$
p	fin pitch, ft
Q	heat-flow rate, Btu/sec
Re	Reynolds number, $wD_h/A\mu$
r	radius, ft

T temperature, $^{\circ}R$

$T_{B,av}$ average blade temperature for any segment, $^{\circ}R$

$T_{c,av}$ average of coolant inlet and outlet temperatures for any segment, $^{\circ}R$

w weight flow, lb/sec

x spanwise distance from blade root, ft

y dimension in direction of fin height, ft

β $\dfrac{h_{c,f,II}T_{c,av,II} + h_{c,f,I}T_{c,av,I}}{k\tau}$, $^{\circ}R/ft$

γ ratio of specific heats

μ viscosity, lb/(sec)(ft)

τ fin thickness, ft

φ $\dfrac{h_{c,f,II} + h_{c,f,I}}{k\tau}$, 1/ft

ω angular velocity, radians/sec

Subscripts:

av average

B blade

c coolant

e effective

in inlet of segment Δx

inlet inlet of coolant passage

f fin

g gas

h blade root (except when used with D_h)

o	outside
out	outlet of segment Δx
outlet	outlet of coolant passage
t	tip
x	spanwise distance from blade root, ft
I	outward-flow passage (see fig. 1(b))
II	inward-flow passage (see fig. 1(b))

APPENDIX B

DERIVATION OF EQUATIONS FOR CALCULATING BLADE-SHELL AND COOLANT TEMPERATURES

From the symmetry of the schematic sketch of the coolant-passage pair shown in figure 7, it can be seen that the heavily shaded inverted T-section (consisting of a chordwise length of blade shell with half of the fin attached at its center) is a representative section of this type of coolant-passage configuration, because its surroundings are identical with that of any other like section. Thus, each representative section is cooled by one-fourth of the total coolant flowing in each passage pair. The derivation of the heat-transfer equations of this investigation is based on this representative section. Inasmuch as the fin attached to a portion of the blade shell serves as an augmenting heat-transfer surface, it is necessary to include the effect of heat conduction from the inner surface of the blade shell into the fin in the derivation of the heat-transfer equation for calculating the blade-shell temperature. Therefore, the temperature distribution in the fin must be derived before the equation for the blade-shell temperature can be derived.

Fin Temperature Distribution

The heat conducted into the fin through its base of attachment to the shell is conducted along its length and removed by convective heat transfer to the coolant flowing in legs I and II. For a differential element of the fin, as illustrated in figure 7 (spanwise conduction is ignored and conductivity normal to y is infinite), the heat balance is

$$-k\tau\,\Delta x\,\frac{dT_f}{dy} = -\,k\tau\,\Delta x\left(\frac{dT_f}{dy} + \frac{d^2T_f}{dy^2}\,dy\right) + h_{c,f,I}(T_f - T_{c,av,I})dy\,\Delta x + h_{c,f,II}(T_f - T_{c,av,II})dy\,\Delta x \tag{B1}$$

This equation reduces to

$$\frac{d^2T_f}{dy^2} - \varphi^2 T_f = -\beta^2 \tag{B2}$$

where

$$\beta^2 = \frac{h_{c,f,I}T_{c,av,I} + h_{c,f,II}T_{c,av,II}}{k\tau}$$

and

$$\varphi^2 = \frac{h_{c,f,I} + h_{c,f,II}}{k\tau}$$

The solution for this equation is

$$T_f = \frac{\beta^2}{\varphi^2} + C_1 e^{\varphi y} + C_2 e^{-\varphi y} \tag{B3}$$

The constants C_1 and C_2 can be evaluated from the boundary conditions. When $y = 0$, $T_f = T_{B,av}$; and, when $y = L/2$, $dT_f/dy = 0$. From these conditions, the temperature distribution in the fin can be expressed as

$$T_f = \frac{\beta^2}{\varphi^2} + \left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right) \frac{e^{\varphi[(L/2)-y]} + e^{-\varphi[(L/2)-y]}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{B4}$$

The evaluation of a heat-transfer coefficient was based on a film temperature, which required knowing a mean fin temperature. The equation for calculating a mean fin temperature is

$$T_{f,av} = \frac{\int_{y=0}^{y=L/2} T_f \, dy}{L/2} \tag{B5}$$

Substituting T_f from equation (B4) into the preceding expression and integrating yield

$$T_{f,av} = \frac{\beta^2}{\varphi^2} + \frac{\left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right)}{\varphi L/2} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{B6}$$

Blade-Shell Temperature

The equation for calculating the blade-shell temperature is derived from a balance of heat flux entering and leaving a segment of the blade shell with an attached fin. This heat-flow summation for a small spanwise segment is

$$h_o p(T_{g,e} - T_{B,av})\Delta x = h_{c,I} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,I})\Delta x + h_{c,II} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,II})\Delta x - k\tau \Delta x \frac{dT_f}{dy} \tag{B7}$$

This equation neglects the variation of the shell temperature between the fins and assumes the shell temperature to be that at the base of the fin. The last term of the equation is the heat conducted from the shell through the base of the fin. Substituting the derivative of equation (B4) with respect to y, when $y = 0$, into equation (B7) yields

$$h_o p(T_{g,e} - T_{B,av}) = h_{c,I} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,I}) + h_{c,II} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,II}) + k\tau\varphi\left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right) \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{B8}$$

Solving equation (B8) for the average shell temperature for the segment yields

$$T_{B,av} = \frac{h_o p T_{g,e} + h_{c,I} T_{c,av,I} \frac{p - \tau}{2} + h_{c,II} T_{c,av,II} \frac{p - \tau}{2} + k\tau\varphi \frac{\beta^2}{\varphi^2} \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}}}{h_o p + h_{c,I} \frac{p - \tau}{2} + h_{c,II} \frac{p - \tau}{2} + k\tau\varphi \frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}}} \tag{1}$$

Coolant Temperature Rise

The temperature rise or drop of the coolant in the passages is due to rotation and convective heat transfer from the blade shell and the fin. The temperature rise from the blade root to any point x due to rotation can be expressed as

$$\Delta T_c = \frac{\omega^2 (r_x^2 - r_h^2)}{2gJc_p} \tag{B9}$$

If the blade span is divided into a number of increments, the temperature rise from the $(n - 1)$ to the n^{th} increment is

$$\Delta T_{c,(n-1) \text{ to } n} = \frac{\omega^2}{2gJc_p} \left\{(r_h + n\,\Delta x)^2 - [r_h + (n - 1)\Delta x]^2\right\} \tag{B10}$$

which reduces to

$$\Delta T_{c,(n-1) \text{ to } n} = \frac{\omega^2}{2gJc_p} [2r_h + \Delta x(2n - 1)]\Delta x \tag{B11}$$

Convective heat transfer from the blade shell to the coolant in leg I can be expressed as

$$Q_{c,I} = h_{c,I} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,I})\Delta x \tag{B12}$$

Similarly, in leg II it is

$$Q_{c,II} = h_{c,II} \frac{p - \tau}{2} (T_{B,av} - T_{c,av,II})\Delta x \tag{B13}$$

Convective heat transfer from one side of the fin to the coolant in leg I can be expressed as

$$Q_{c,f,I} = \Delta x \int_{y=0}^{y=L/2} h_{c,f,I}(T_f - T_{c,av,I})dy \tag{B14}$$

Similarly, for leg II

$$Q_{c,f,II} = \Delta x \int_{y=0}^{y=L/2} h_{c,f,II}(T_f - T_{c,av,II})dy \tag{B15}$$

Substituting equation (B4) into equations (B14) and (B15) and integrating over the limits indicated yield

$$Q_{c,f,I} = h_{c,f,I}\,\Delta x\,\frac{L}{2}\left(\frac{\beta^2}{\varphi^2} - T_{c,I}\right) + h_{c,f,I}\,\Delta x\,\frac{T_{B,av} - \frac{\beta^2}{\varphi^2}}{\varphi}\,\frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{B16}$$

$$Q_{c,f,II} = h_{c,f,II}\,\Delta x\,\frac{L}{2}\left(\frac{\beta^2}{\varphi^2} - T_{c,II}\right) + h_{c,f,II}\,\Delta x\,\frac{T_{B,av} - \frac{\beta^2}{\varphi^2}}{\varphi}\,\frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{B17}$$

The temperature of the coolant leaving any segment of leg I can be obtained by adding to the coolant temperature at the inlet of the segment the temperature rise due to rotation (eq. (B11)) and the temperature rise due to convective transfer from the shell and the fin (eqs. (B12) and (B16), respectively, divided by $w_c c_p/4$). These equations are divided by one-fourth of the total coolant flow per passage pair for the reasons indicated at the beginning of this appendix.

$$T_{c,I,out} = T_{c,I,in} + \frac{\omega^2\,\Delta x}{2gJc_p}\,[2r_h + \Delta x(2n - 1)] + 2h_{c,I}\,\Delta x\,\frac{(p - \tau)(T_{B,av} - T_{c,av,I})}{w_c c_p} +$$
$$\frac{2h_{c,f,I}L\,\Delta x\left(\frac{\beta^2}{\varphi^2} - T_{c,av,I}\right)}{w_c c_p} + \frac{4h_{c,f,I}\,\Delta x\left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right)}{\varphi w_c c_p}\,\frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{6}$$

Similarly, the temperature of the coolant entering leg II for the segment can be obtained by using equations (B11), (B13), and (B17). However, in this case the terms representing temperature change due to convective heat transfer are subtracted from the segment outlet temperature plus the temperature rise due to rotation, because coolant-flow direction is opposite to the x-direction:

$$T_{c,II,in} = T_{c,II,out} + \frac{\omega^2\,\Delta x}{2gJc_p}\,[2r_h + \Delta x(2n - 1)] - 2h_{c,II}\,\Delta x\,\frac{(p - \tau)(T_{B,av} - T_{c,av,II})}{w_c c_p} -$$
$$\frac{2h_{c,f,II}L\,\Delta x\left(\frac{\beta^2}{\varphi^2} - T_{c,av,II}\right)}{w_c c_p} - \frac{4h_{c,f,II}\,\Delta x\left(T_{B,av} - \frac{\beta^2}{\varphi^2}\right)}{\varphi w_c c_p}\,\frac{e^{\varphi L/2} - e^{-\varphi L/2}}{e^{\varphi L/2} + e^{-\varphi L/2}} \tag{7}$$

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TABLE I. - COMPARISON OF COOLANT-FLOW REQUIREMENTS WHEN FIN THICKNESS AND FIN HEIGHT ARE VARIED

[Coolant, hydrogen; coolant-inlet temperature, 250° R; turbine-inlet temperature, 3000° R; blade-root stress, 50,000 psi; stress-ratio factor, 1.5; fin pitch, 0.105 in.; flight altitude, 50,000 ft; flight Mach number, 2.5.]

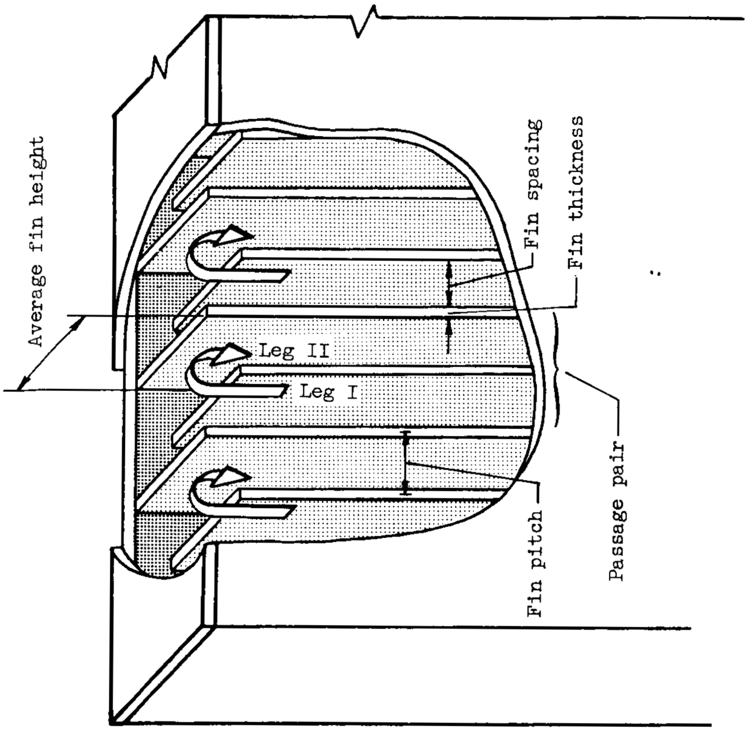
Geometry	Average fin height, in.	Fin thickness, in.	Fin spacing, in.	Coolant flow per passage pair, lb/sec	$\frac{\text{Fin height}}{\text{Fin thickness}}$, L/τ
A	0.20	0.005	0.100	0.000612	40
B	.20	.010	.095	.000548	20
C	.15	.005	.100	.000528	30
D	.15	.010	.095	.000489	15

TABLE II. - COOLANT-FLOW AND PRESSURE REQUIREMENTS

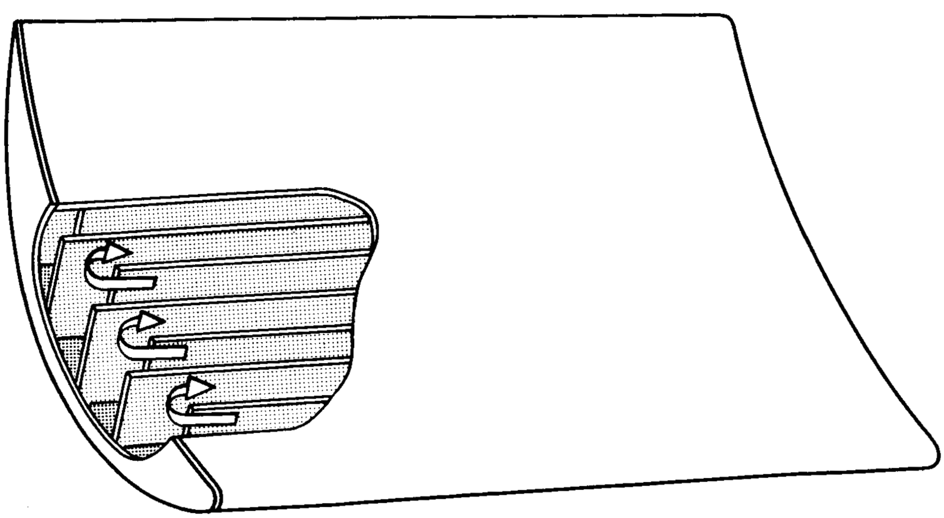
[Turbine-inlet temperature, 3000° R; blade-root stress, 50,000 psi; stress-ratio factor, 1.5; fin thickness, 0.005 in.; fin spacing, 0.100 in.; flight Mach number, 2.5.]

Gas-to-blade heat-transfer coefficient, h_o, Btu/(sec)(sq ft)(°R)	Coolant inlet temperature, $T_{c,I,inlet}$, °R	Required coolant flow per blade, w_c, lb/sec	Coolant inlet static pressure, $P_{c,I,inlet}$, lb/sq ft	Coolant static pressure at blade tip, $P_{c,t}$, lb/sq ft	Coolant outlet static pressure, $P_{c,II,outlet}$, lb/sq ft
(a) Hydrogen					
0.0580	100	0.00778	3,000 6,000 9,000 20,000	3,495 7,047 10,586 23,567	3,308 6,802 10,253 22,874
	250	0.00734	3,000 6,000 9,000 20,000	3,170 6,441 9,688 21,569	2,998 6,277 9,489 21,194
	500	0.00751	3,000 6,000 9,000 20,000	3,006 6,199 9,348 20,850	2,778 6,027 9,161 20,541
	750	0.00895	3,000 6,000 9,000 20,000	2,845 6,069 9,200 20,584	2,451 5,838 8,976 20,265
	1000	0.01233	3,000 6,000 9,000 20,000	(a) 5,878 9,045 20,424	(a) 5,468 8,711 20,063
0.0213	100	0.00115	3,000 6,000 9,000	3,285 6,580 9,924	3,290 6,600 9,957
	250	0.00137	3,000 6,000 9,000	3,169 6,344 9,506	3,159 6,337 9,497
	500	0.00208	3,000 6,000 9,000	3,111 6,238 9,364	3,069 6,175 9,274
	750	0.00282	3,000 6,000 9,000	3,068 6,169 9,264	3,004 6,082 9,144
	1000	0.00384	3,000 6,000 9,000	3,024 6,119 9,198	2,928 6,012 9,059
(b) Methane					
0.0580	250	0.0449	3,000 6,000 9,000 20,000	4,994 10,214 15,381 34,267	3,409 7,638 11,655 26,176
	500	0.0449	3,000 6,000 9,000 20,000	3,672 7,995 12,172 27,218	1,913 6,286 9,873 22,382

[a]No solution.

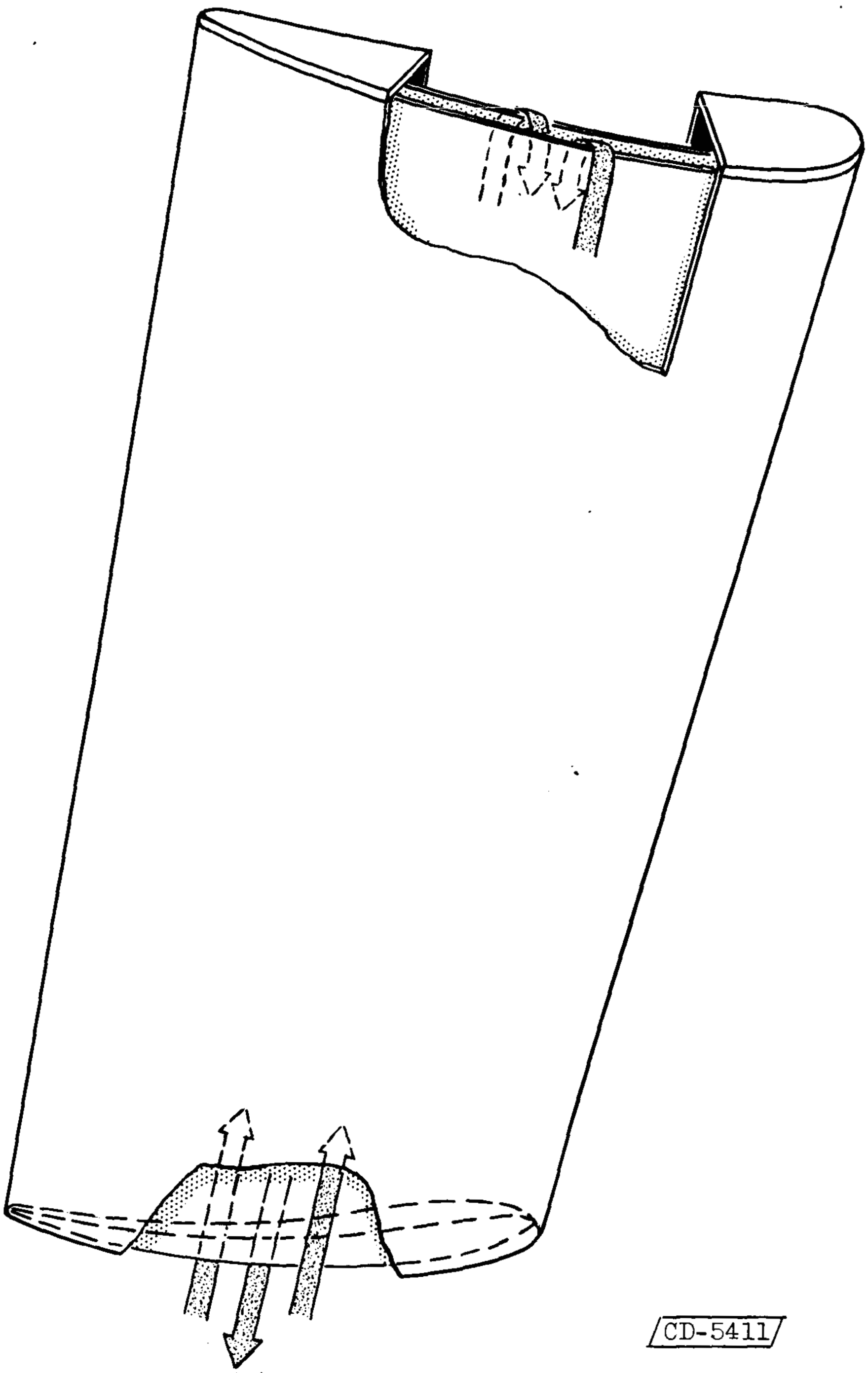


(b) Schematic mean geometry of blade used in this investigation.



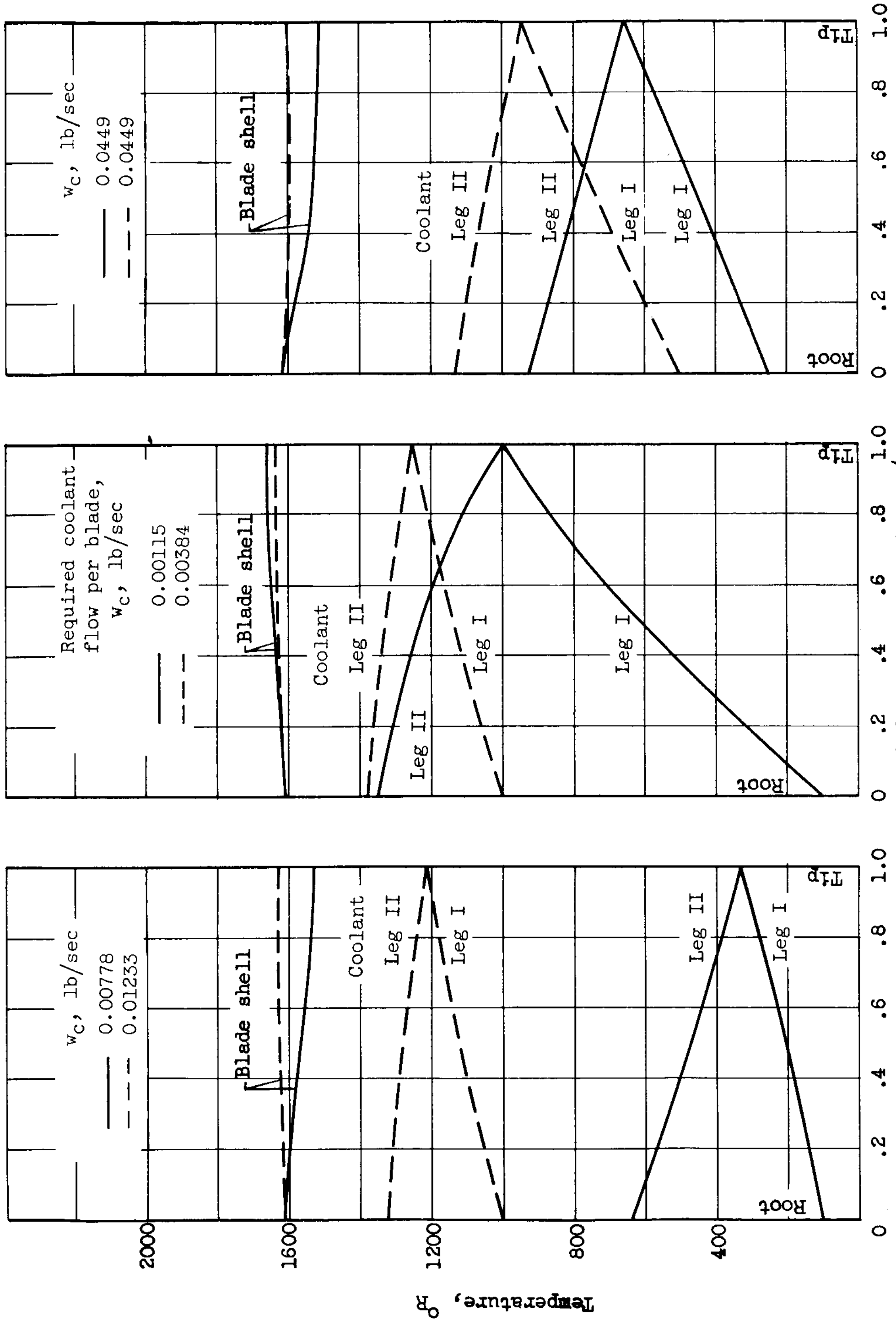
(a) Blade used in this investigation.

Figure 1. - Return-flow type of coolant-passage configurations for turbine rotor blades.



(c) Blade used in reference 5.

Figure 1. - Concluded. Return-flow type of coolant-passage configurations for turbine rotor blades.



(a) Hydrogen; flight altitude, 50,000 feet.

(b) Hydrogen; flight altitude, 80,000 feet.

(c) Methane; flight altitude, 50,000 feet.

Figure 2. - Spanwise blade-shell and coolant temperature distributions. Effective gas temperature, 2765° R; stress-ratio factor, 1.5; flight Mach number, 2.5.

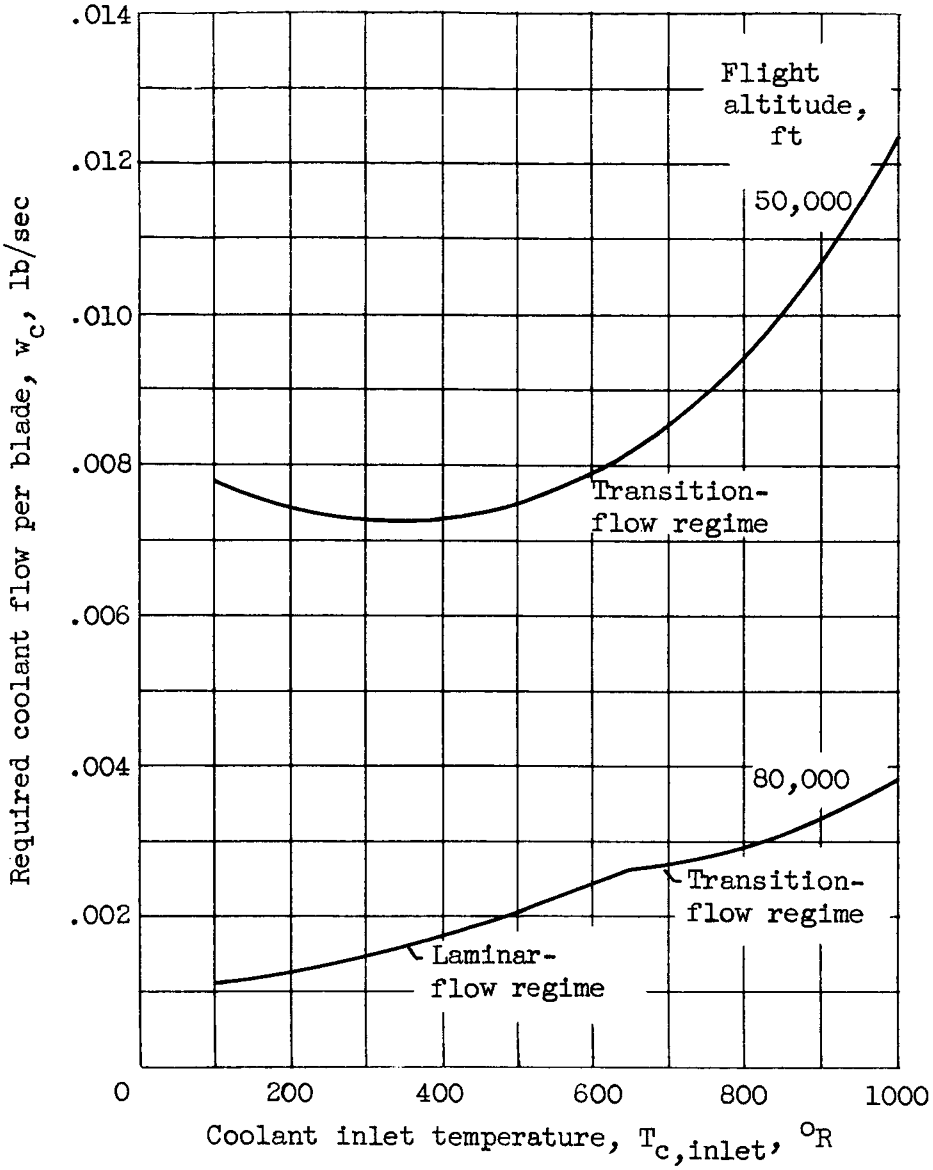


Figure 3. - Variation of coolant-flow requirements with coolant inlet temperature. Coolant, hydrogen; effective gas temperature, 2765° R; stress-ratio factor, 1.5; flight Mach number, 2.5.

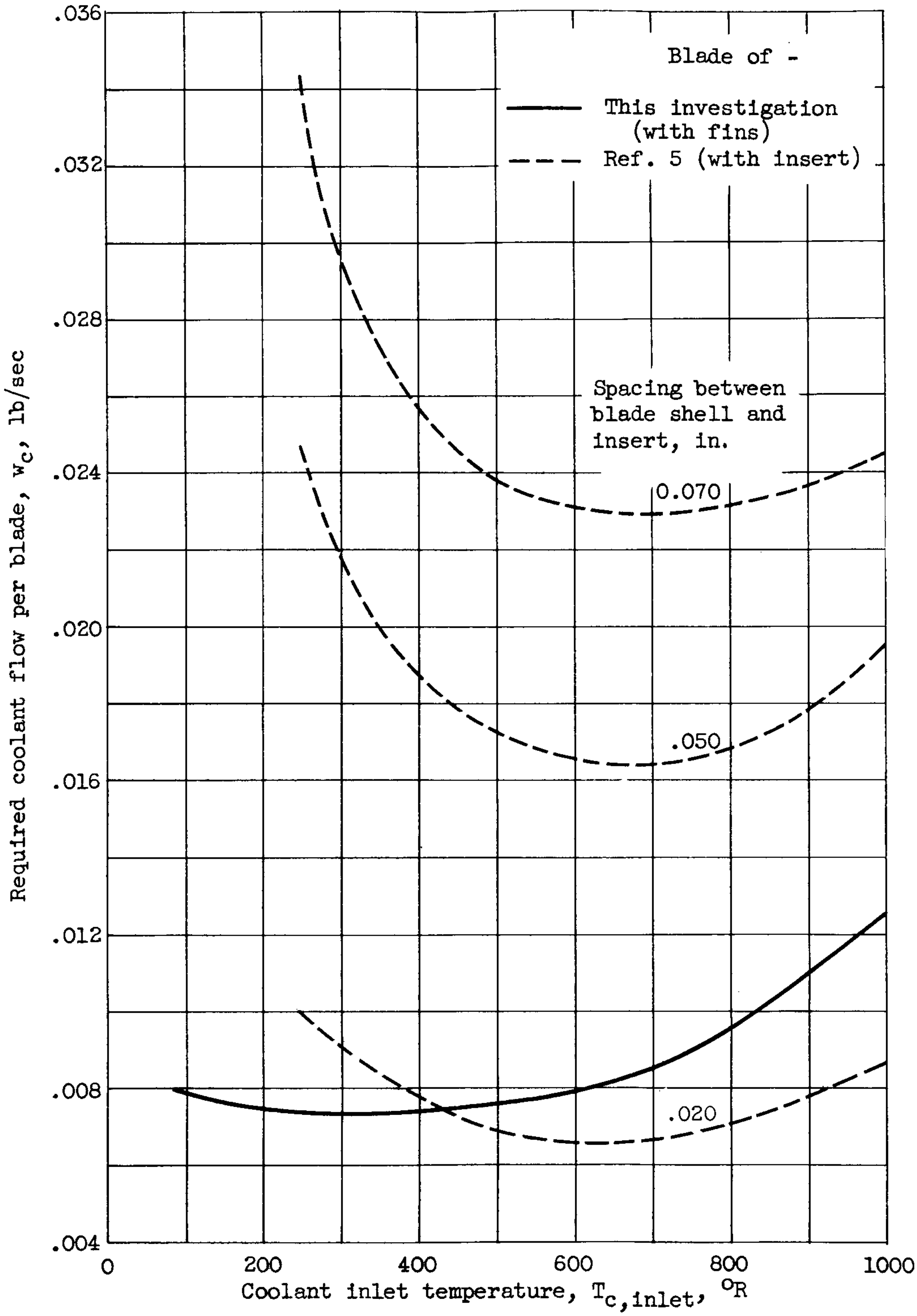


Figure 4. - Comparison of coolant-flow requirements for return-flow blades with insert (ref. 5) and with fins. Coolant, hydrogen; effective gas temperature, 2765° R; flight altitude, 50,000 feet; flight Mach number, 2.5.

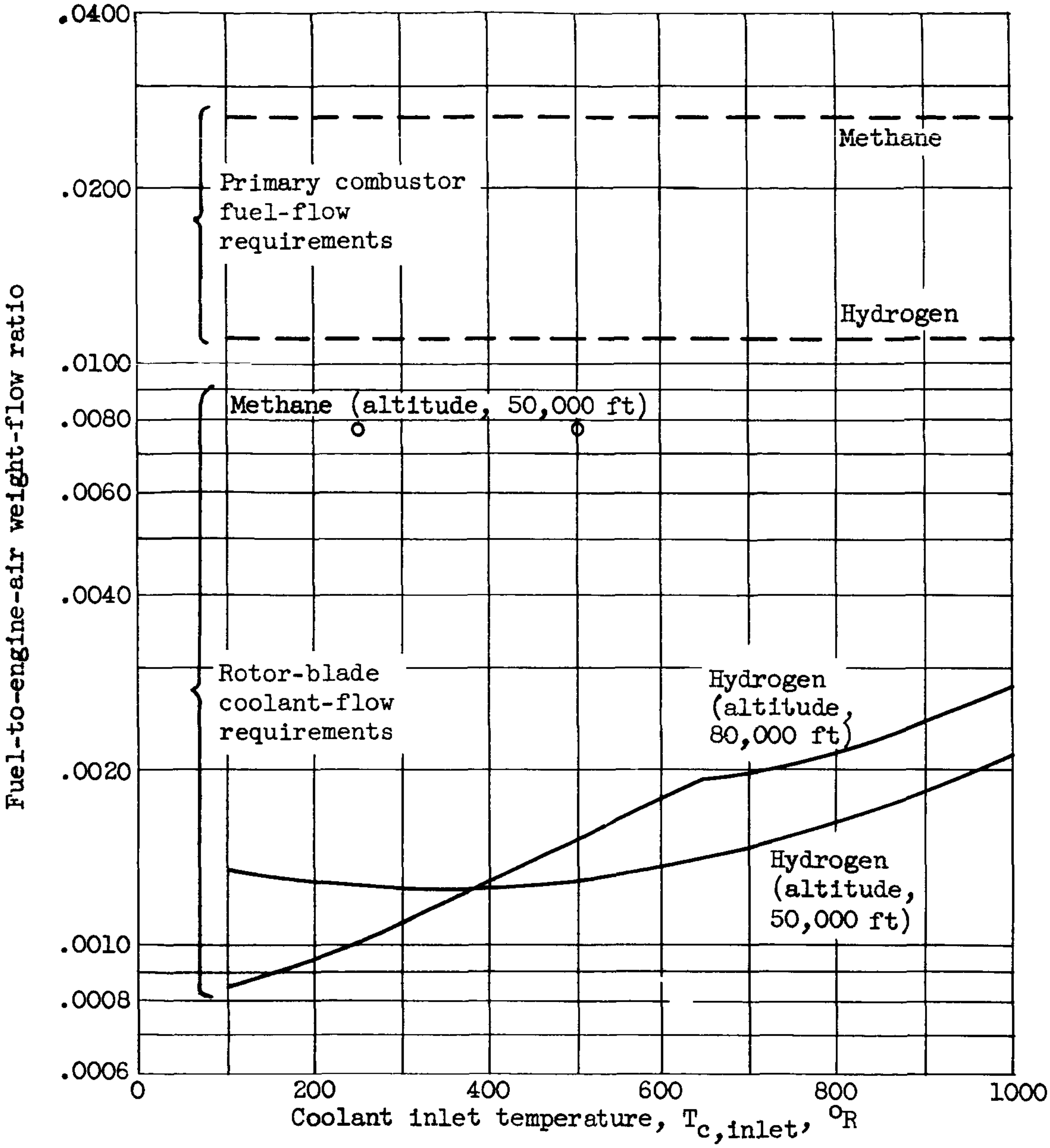


Figure 5. - Comparison of primary combustor fuel-flow requirements with those of rotor-blade coolant-flow requirements. Effective gas temperature, 2765° R; turbine-inlet temperature, 3000° R; flight Mach number, 2.5.

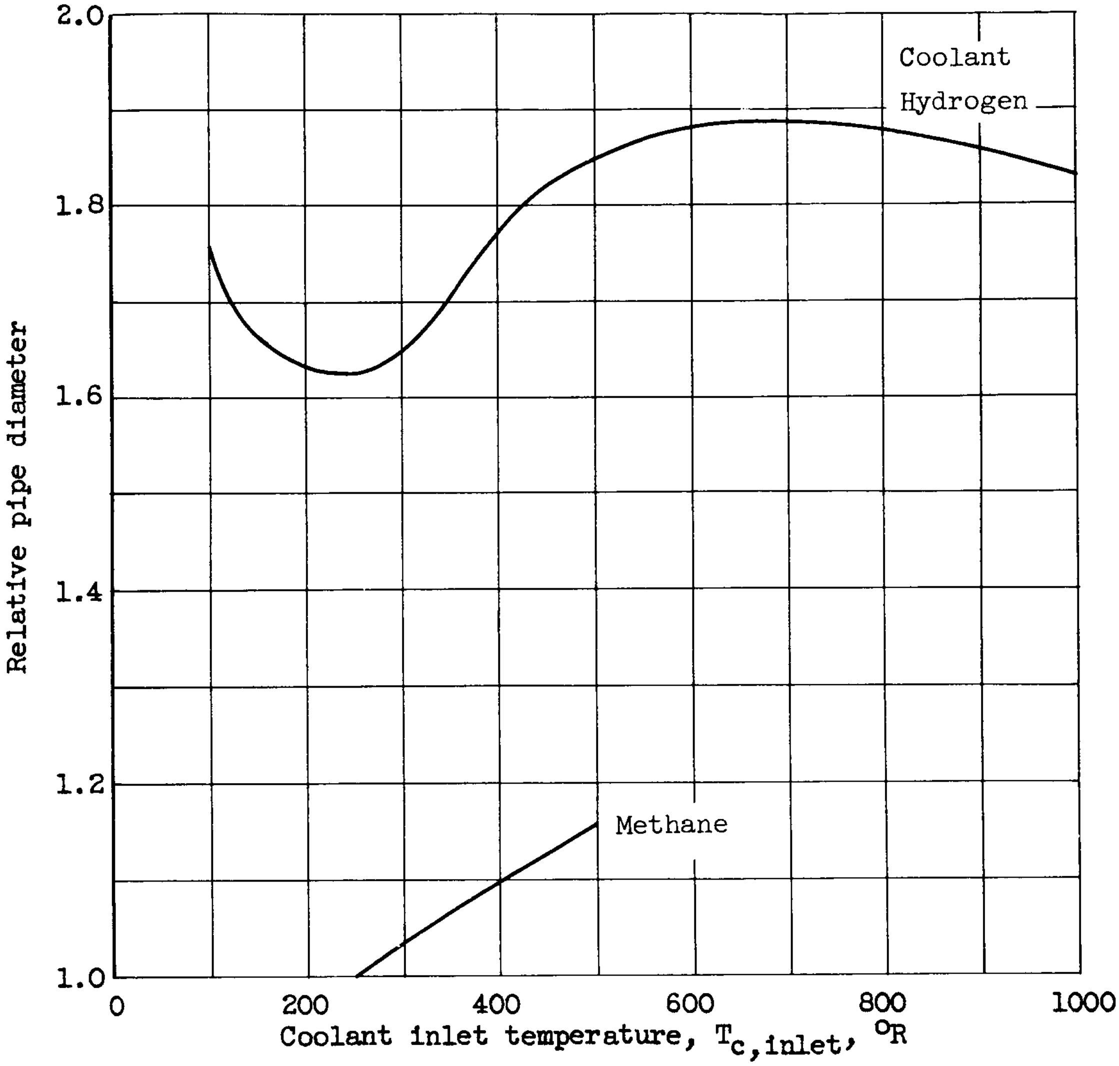


Figure 6. - Variation of relative pipe diameter with coolant inlet temperature for constant pipe pressure loss. Gas-to-blade heat-transfer coefficient, 0.058 Btu/(sec)(sq ft)(^{o}F); flight altitude, 50,000 feet; flight Mach number, 2.5.

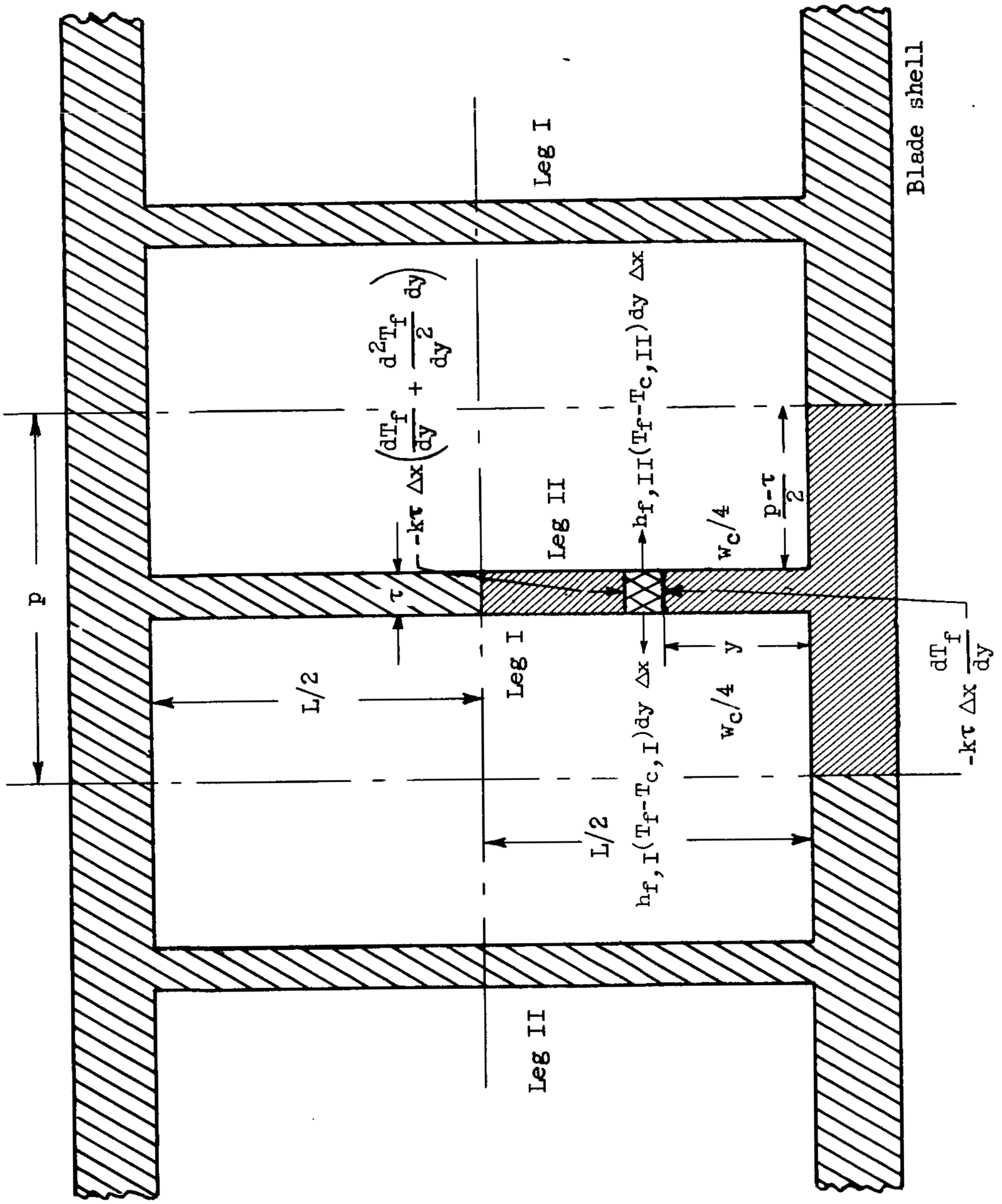


Figure 7. - Coolant-passage pair showing geometry and dimensions pertinent to derivation of equations for calculating blade-shell and coolant temperatures.

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